ORIGINAL SCIENTIFIC PAPER OPEN ACCESS

Analysis of the performance of the shell and tube heat exchanger: influence of pipe layout and diameter

Stefan Adžić[1*], Dragiša Šimunović[1], Branko Radičević[1]

[1]University of Kragujevac, Faculty of Mechanical and Civil Engineering, Kraljevo, Serbia

ARTICLE INFO

* **Correspondence:** stefan_adzic@yahoo.com

DOI: 10.5937/engtoday2300013A

UDC: 621(497.11) **ISSN:** 2812-9474

Article history: Received 20 August 2023; Revised 2 October 2023; Accepted 10 October 2023

ABSTRACT

Heat exchangers are devices that are used in numerous areas of industry with the most important role in process technology. The growing need for maximum energy utilization contributed to the very rapid development of these devices. Of all types of heat exchangers, shell and tube exchangers are the most widely used in process technology. The paper analyzes the performance of a shell and tube heat exchanger of known geometric characteristics and rated power. Numerically and by simulation of fluid flow in the SolidWorks software package, a comparative performance analysis was performed depending on the different layout and for the values of the outer diameter of the pipe 33,7x2 mm and 60,3x2,7 mm for the same external dimensions of the heat exchanger. The obtained results show that the heat exchanger with a circular arrangement of pipes measuring 60,2x2,7 mm has the best performance, taking into account the achieved power of the exchanger.

KEYWORDS

Shell and tube heat exchanger, Simulation, Comparative performance analysis

1. INTRODUCTION

The development of industrial needs was accompanied by the development of heat exchangers, so that the basic structural forms were known in the thirties of the last century, when the development of standards in this area began. Heat exchangers are devices in which heat is exchanged between warmer and colder fluids without a fluids mixture [1,2]. In indirect heat exchange, the fluids are separated by a heating surface, on one side of the surface there is one fluid, and on the other side another fluid. In most cases, there is no forced movement of fluid in heat exchangers, but the flow of fluid in the process is achieved naturally based on the difference in densities. The application of heat exchangers is widespread in various technological processes. Heat exchangers are used in the processes of heating, cooling, condensation, evaporation, crystallization, melting, as well as in heating, cooling and air conditioning systems.

The main problem in the application of these devices is the complexity of determining the output parameters or determining the optimal construction that provides the required parameters. The difficulties that arise in the selection of these devices are the result of complex heat exchange processes [3]. Namely, the visual appearance, dimensions, and even the available data on the basic geometry are not enough to know about the capabilities of the device. Increased exchange, which most often occurs as a parameter of the size of the device, is not sufficient information for making a decision on the use of the device. In the process of applying these devices, in addition to knowing the way heat spreads, it is necessary to know the technological process that is carried out with the help of heat exchangers, as well as the configuration of the flow and the geometrical influences on obtaining the optimal construction. Overcoming potential problems is achieved by thermal, current and mechanical calculations of the heat exchanger, as well as by optimizing the construction of the device.

There are several different categorizations and divisions of exchangers, for example according to construction, method of heat transfer, area of application, types of fluids, flow directions, number of exchanges, number of fluid inlets and outlets, materials used, etc. According to the method of heat transfer from one body to another, exchangers can be divided into regenerative heat exchangers, recuperative heat exchangers, heat exchangers with an intermediate heat exchanger, and contact heat exchangers.

2. RECUPERATIVE HEAT EXCHANGERS

Recuperative heat exchangers are the most commonly used devices in the process industry and are made in a large number of different shapes using a variety of materials. In general, they can be divided into three groups: plate heat exchangers, non-metal heat exchangers and tubular heat exchangers. Shell and tube heat exchangers are one of the most common heat exchangers in process engineering, thermoenergetics and thermotechnics. Their role is multiple, and they can be used as heaters and coolers, condensers, evaporators and vaporizers.

The first step in the design of shell and tube heat exchangers is to define which of the fluids flows from the inside and which from the outside of the tube, i.e. inside the casing. The flow pattern depends on the dynamic and thermodynamic aspects of the fluid and on the allowable pressure drop. The fluid chosen to flow in the pipe should be at a higher pressure, tend to be more dirty, but require expensive materials for production due to the corrosive effect. Fluids that have a low heat transfer coefficient should be on the side of the envelope. In the paper, the operation of the shell and tube heat exchanger was analyzed from the aspect of maximum heat exchange depending on the different diameters and arrangement of pipes in the pipe bundle. In all variant solutions, the length of the active part of the heat exchanger is 3750 mm, the diameter of the device is 1000 mm, while the nominal power of the exchanger is 560 kW.

3. PIPE ARRANGEMENT AT SHELL AND TUBE HEAT EXCHANGERS

In practice, there are several different pipe layouts in shell and tube heat exchangers, while three basic pipe layouts are most often applied: triangular (by the vertices of an equilateral triangle), corridor or rectangular (by the vertices of a rectangle) and checkered or circular (by concentric circles).

The paper analyzed two pipe layouts with two different pipe diameter values in the pipe bundle: 33,7x2 mm and 60,3x2,7 mm. The numbers of pipes depending on the layout and dimensions of the diameter are obtained graphically with the limitation of the minimum distance between the pipes which is $(1,25 \div 1,5) \cdot d_s$. The pipe numbers for both analyzed cases are shown in the following table.

Table 1: Number of pipes depending on the layout and outer diameter

Pipe diameter [mm]	**Corridor layout**	**Circular layout**
33,7x2	277	294
60,3x2,7	104	142

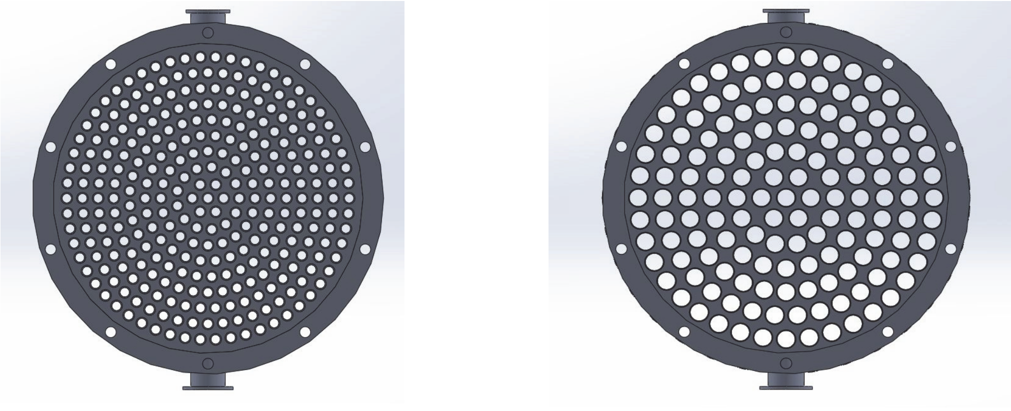

Figure 1: Graphic representation of pipes in a circular arrangement

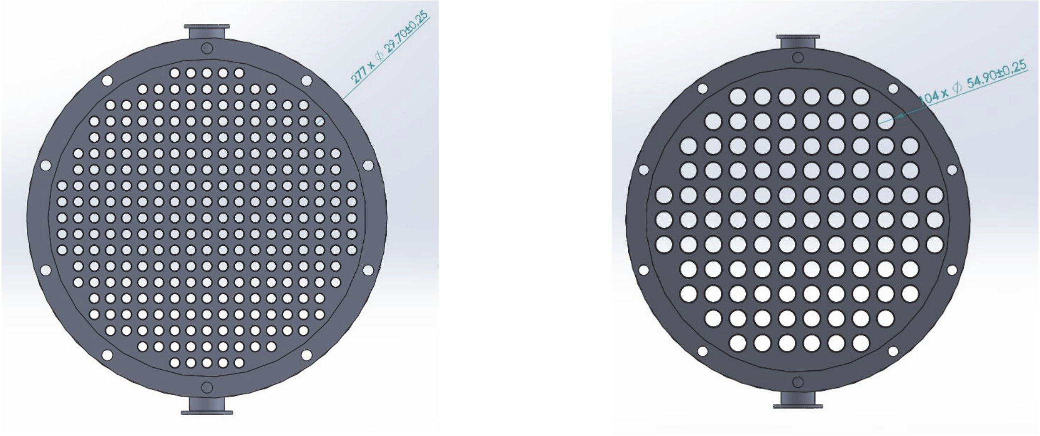

Figure 2: Graphic representation of pipes in a rectangular layout

4. MODELING AND CALCULATION

The starting equation for dimensioning the heat exchanger is the energy balance equation. It is necessary to know the following quantities in order to perform the calculation, that is, the sizing of the heat exchanger itself: mass flows, inlet temperatures and outlet temperature of one of the fluids. The energy balance equation is:

$$Q = \dot{m}_1 \cdot c_{p1} \cdot \left(t_{1\,in} - t_{1\,out}\right) = \dot{m}_2 \cdot c_{p2} \cdot \left(t_{2\,out} - t_{2\,in}\right) = k \cdot A_{exch} \cdot \Delta t_m \tag{1}$$

In equation (1) are: $\dot{m}_1$ - mass flow rate of the fluid 1, $c_{p,1}$-specific heat capacity of the fluid 1, $t_{1\,in}$ - inlet temperature of the fluid 1, t_{1out} - outlet temperature of the fluid 1, $\dot{m}_2$ -mass flow rate of the fluid 2, c_{p2} - specific heat capacity of the fluid 2, $t_{2\,in}$ - inlet temperature of the fluid 2, t_{2out} - outlet temperature of the fluid 2.

From equation (2), we can determine the mean logarithmic temperature difference for a counterflow heat exchanger, while the correction factor ε is obtained from [4, 5, 6]:

$$\Delta t_{mlog} = \varepsilon \cdot \frac{\left(t_{1\,in} - t_{2\,out}\right) - \left(t_{1\,out} - t_{2\,in}\right)}{ln\left(\dfrac{t_{1\,in} - t_{2\,out}}{t_{1\,out} - t_{2\,in}}\right)} \tag{2}$$

Based on the known values of the inlet and outlet temperatures of both fluids, the value of the diameter of the pipes and the exchanger shell, the heat transfer coefficient is obtained from equation (3):

$$k = \frac{1}{\left(\dfrac{1}{\alpha_1} + R_{f1}\right)\dfrac{A_2}{A_1} + R_{wall} + \dfrac{1}{\alpha_2} + R_{f2}} \tag{3}$$

In equation (1) are: α_i-heat transfer coefficients, A_i-heat transfer areas, R_{f1} - fouling resistances, R_{wall} -thermal resistance of the separating wall [7]. The heat resistance coefficients due to contamination were adopted from [8]:

$$R_{f1} = 0{,}2 \cdot 10^{-3}\,\frac{m^2K}{W} \quad \text{and} \quad R_{f2} = 0{,}1 \cdot 10^{-3}\,\frac{m^2K}{W} \tag{4}$$

Convective heat transfer coefficient is obtained by the formula:

$$\alpha_i = \frac{Nu_i \cdot \lambda_i}{d_{e,i}} \tag{5}$$

where are: λ_i – the coefficient of the conduction, $d_{e,i}$ – the equivalent diameter, Nu_i – Nusselt number [8].

Dimensionless numbers, required for determination of the heat transfer coefficient, were calculated from (6) and (7).

$$Nu_i = 0{,}21 \cdot \mathrm{Re}_i^{0{,}6} \cdot Pr_i^{\frac{1}{3}} \tag{6}$$

$$\mathrm{Re}_i = \frac{w_i \cdot d_{e,i} \cdot \rho_i}{\mu_i} \tag{7}$$

where are: w_i – the fluid velocity, ρ_i – the fluid density, μ_i – the coefficient of dynamic viscosity of the fluids, Re_i – Reynolds number, Pr_i – Prandtl number, where is $i = 1$ - for fluid 1, $i = 2$ – for fluid 2.

Based on previous, the outlet temperature of the fluids and heat transfer coefficient were determined for the given input parameters. The specific heat capacities of both fluids were adopted based on the mean temperatures from [4, 5, 6].

Table 2: Values of heat transfer coefficient and heat exchange surface obtained by analytical calculation

<table>
<tr><th>Fluid</th><th colspan="6">Fluid 1 - water (warmer)</th><th colspan="6">Fluid 2 - water (colder)</th></tr>
<tr><td rowspan="2">Parameters</td><td colspan="2">$\dot{m}_1$ [kg/s]</td><td colspan="2">t_{1in} [°C]</td><td colspan="2">t_{1out} [°C]</td><td colspan="2">$\dot{m}_2$ [kg/s]</td><td colspan="2">t_{2in} [°C]</td><td colspan="2">t_{2out} [°C]</td></tr>
<tr><td colspan="2">1,67</td><td colspan="2">180</td><td colspan="2">101,74</td><td colspan="2">2,9</td><td colspan="2">40</td><td colspan="2">86,13</td></tr>
<tr><td>Pipe arrangement</td><td colspan="6">Circular</td><td colspan="6">Corridor</td></tr>
<tr><td>Pipe diameter [mm]</td><td colspan="3">33,7x2</td><td colspan="3">60,3x2,7</td><td colspan="3">33,7x2</td><td colspan="3">60,3x2,7</td></tr>
<tr><td>Exchange area A [m2]</td><td colspan="3">116,72</td><td colspan="3">100,88</td><td colspan="3">109,97</td><td colspan="3">73,88</td></tr>
<tr><td>Heat transfer coefficient k [W/m^{2}K]</td><td colspan="3">62,56</td><td colspan="3">72,39</td><td colspan="3">66,40</td><td colspan="3">98,84</td></tr>
</table>

5. MODEL VALIDATION

By comparing the results obtained on the basis of analytical calculations and the results obtained by simulations for all four analyzed cases, validation was performed. The input parameters in the simulation model are identical to the parameters in the theoretical model. The outlet temperatures of both fluids were the parameters used for model validation. Given that the difference between the temperatures at the outlet of the colder and warmer fluid compared to the theoretical model in all four cases is a maximum of 8,3%, the simulation model is able to predict the temperature exchange in the heat exchanger.

6. FLOW ANALYSIS USING SOLIDWORKS FLOW SIMULATION

On the basis of pre-defined input parameters, models were created for all four cases on which simulation of fluid flow inside the exchanger was carried out and further analyzes of the obtained results were carried out. For all proposed designs of heat exchangers, the following parameters were taken into account: the system is adiabatically isolated, the fluid flow is countercurrent, the fluids entering the apparatus have a fully developed flow, the heat exchanger is made of steel, the number of inlets and outlets for the hotter and colder fluid is the same in all variants of the solution. In order to achieve the most uniform flow of fluid in the tube register and more intense heat exchange, the number and arrangement of tubes was optimized. Variant solutions imply changes in the diameter of the pipes in the heat exchanger itself, as well as different arrangements of the pipes. By simulating the fluid flow when applying these variant solutions, results were obtained that show that the most intensive heat exchange is achieved in the case of heat exchangers with a circular arrangement of pipes and a diameter of 60,3x2,7 mm. The deviation of the output temperature of the warmer fluid is only 2,49 percent compared to the theoretically obtained results.

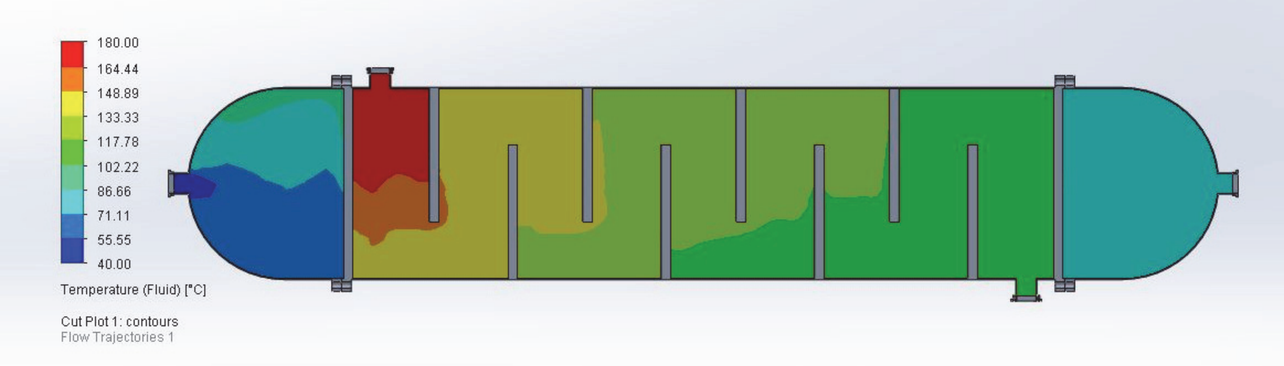

Figure 3: Temperature field for the exchanger model with a rectangular arrangement of pipes with a diameter of 60,3x2,7 mm obtained by simulation

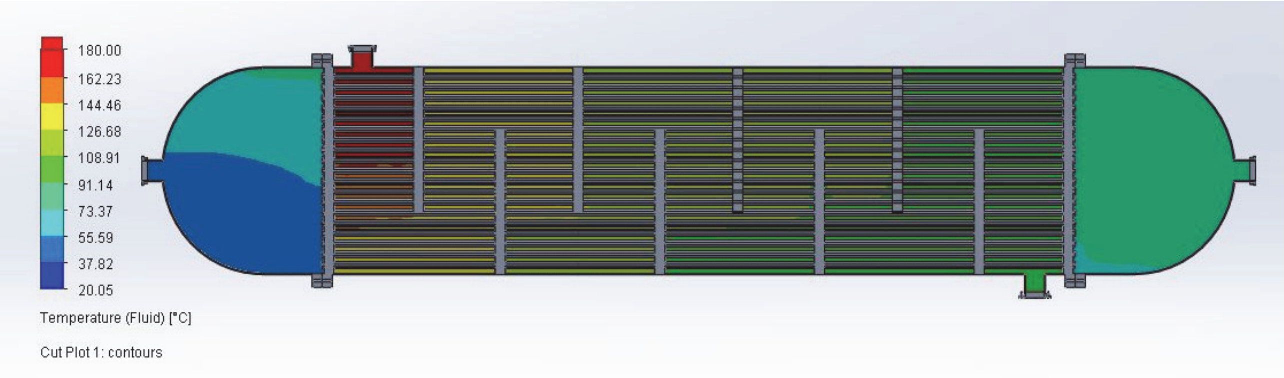

Figure 4: Temperature field for the model of the exchanger with a rectangular arrangement of pipes with a diameter of 33,7x2 mm obtained by simulation

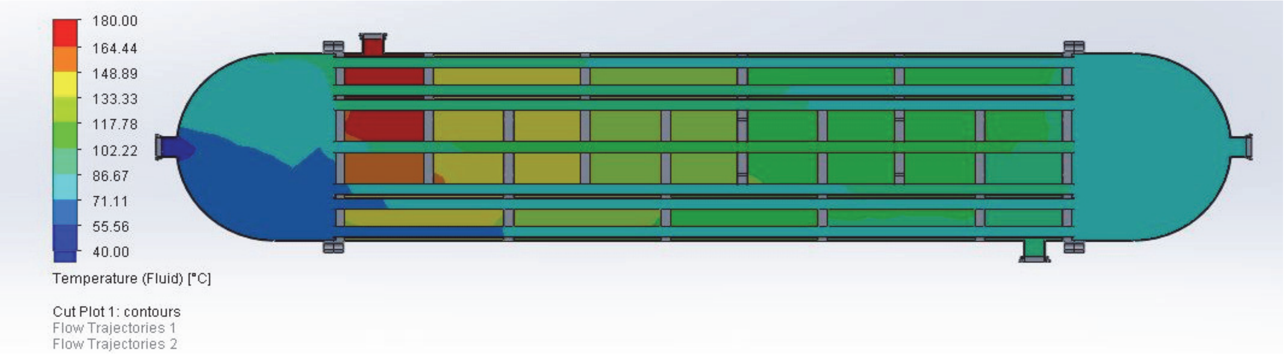

Figure 5: Temperature field for the exchanger model with a circular arrangement of tubes with a diameter of 60,3x2,7 mm obtained by simulation

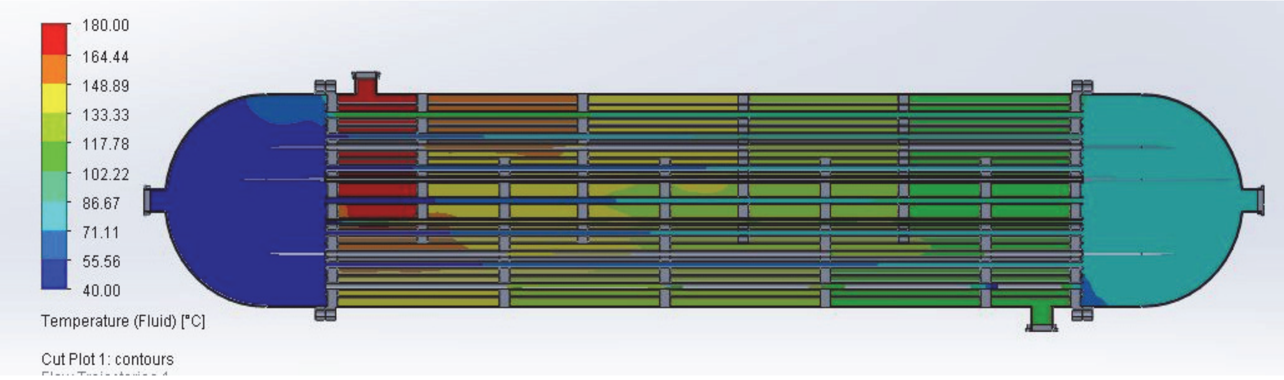

Figure 6: Temperature field for the exchanger model with a circular arrangement of pipes with a diameter of 33,7x2 mm obtained by simulation

7. RESULTS AND ANALYSIS

The following table shows the values of the inlet and outlet temperatures of the fluid in the simulation of the operation of the shell and tube heat exchanger, in which four different geometry optimization solutions were made. Based on the simulation and analysis of the fluid flow, it is concluded that the most adequate optimization solution is the one in which the largest temperature difference is obtained, that is, the largest exchanged amount of heat compared to the theoretical model. In all optimization solutions, the mass flow values of colder and warmer fluids and their inlet temperatures didn't change.

Table 2: Inlet and outlet temperatures depending of type construction obtained by simulation

Fluid		Fluid 1 - water (warmer)			Fluid 2 - water (colder)		
Pipe arrangement		Circular					
Parameters		$\dot{m}_1$ [kg/s]	t_{1in} [°C]	t_{1out} [°C]	$\dot{m}_2$ [kg/s]	t_{2in} [°C]	t_{2out} [°C]
Pipe diameter [mm]	33,7x2	1,67	180	109,40	2,9	40	79,67
	60,3x2,7	1,67	180	99,21	2,9	40	80,10
Pipe arrangement		Corridor					
Parameters		$\dot{m}_1$ [kg/s]	t_{1in} [°C]	t_{1out} [°C]	$\dot{m}_2$ [kg/s]	t_{2in} [°C]	t_{2out} [°C]
Pipe diameter [mm]	33,7x2	1,67	180	98,75	2,9	40	80,72
	60,3x2,7	1,67	180	105,77	2,9	40	79,01

Table 3: Deviations of power, input and output temperatures from the theoretical model depending on the type of construction obtained by simulation

Pipe arrangement		Circular			
Deviations in relation to the values obtained by theoretical calculation [%]		t_{1out}	t_{2out}	Q	Q_2 / Q_1
Pipe diameter [mm]	33,7x2	-7,53	7,50	9,60	**95,11**
	60,3x2,7	2,49	7,00	**-3,16**	84,25
Pipe arrangement		Corridor			
Deviations in relation to the values obtained by theoretical calculation [%]		t_{1out}	t_{2out}	Q	Q_2 / Q_1
Pipe diameter [mm]	33,7x2	2,94	6,28	-3,73	85,08
	60,3x2,7	-3,96	8,27	5,05	89,04

8. CONCLUSION

The paper presents an analysis of the geometric characteristics in the operation of a shell and tube heat exchanger with two floating heads. Simulations were performed in the "Solidworks" software package for four variant solutions. In all four variant solutions, complete heat exchange between colder and warmer fluid was not achieved, which shows that in the analyzed constructions, the exchange surface must be increased by adding partitions or increasing the number of pipes in the pipe bundle. Respecting the theoretically defined pipe diameter distances, it is not possible to significantly increase the surface area for heat exchange by adding pipes, so changing the diameter of the pipe and thus increasing the surface area for exchange remains as a proposed solution. The most optimal solution of the four analyzed cases is a construction with a circular arrangement of pipes with a diameter of 60,3x2,7 mm and its deviation taking into account the power of the exchanger it can achieve is 3,16%, while taking into account the amount of heat exchanged between the fluids, the most optimal construction would was with a circular arrangement and a diameter of 33,7x2 mm with an achieved exchange of 95,11%.

ACKNOWLEDGMENT

This research has been supported by the Serbian Ministry of the Education, Science and Technology Development under grant 451-03-68/2022-14/200108.

REFERENCES

[1] S. H. Gawande, S. D. Wankhede, R. N. Yerrawar, V. J. Sonawane and U. B. Ubarhande, “Design and Development of Shell & Tube Heat Exchanger for Beverage”, Modern Mechanical Engineering, Vol. 2(4), pp. 121-125, http://dx.doi.org/10.4236/mme.2012.24015, (2012)

[2] L-Y. Chen, V. S. K. Adi and R. Laxmidewi, “Shell and tube heat exchanger flexible design strategy for process operability”, Case Studies in Thermal Engineering, Vol. 37, 102163, https://doi.org/10.1016/j.csite.2022.102163, (2022)

[3] M. A. Tomić, “Experimental and numerical investigation of thermal and fluid flow processes in package of perforated plates”, PhD thesis, University of Niš, Niš (Serbia), (2015)

[4] B. Jaćimović and S. Genić, “Toplotne operacije i aparati – Deo 1: Rekuperativni razmenjivači toplote”, Mašinski fakultet, Beograd (Serbia), (2016)

[5] S. Genić, B. Jaćimović, M. Jarić and N. Budimir, “Svojstva procesnih fluida”, Savez mašinskih i elektrotehničkih inženjera i tehničara Srbije, Beograd (Serbia), (2014)

[6] B. Jaćimović, S. Genić, M. Nađ and J. Laza, “Problemi iz toplotnih operacija i aparata”, SMEITS i Mašinski fakultet, Beograd (Serbia), (1996)

[7] H. Müller-Steinhagen, “C4 Fouling of Heat Exchanger Surfaces” in VDI Heat Atlas, VDI-Buch. Springer, Berlin (Germany), https://doi.org/10.1007/978-3-540-77877-6_7, (2010)

[8] M. Marašević, N. Bogojević, N. Stojić, S. Adžić and D.Šimunović, “Optimization of compact heat exchanger“, Proceedings of X International Conference “Heavy Machinery HM 2021”, pp. 497-502, (2021)